DEFENCE RESEARCH BOARD OF CANADA
IN ASSOCIATION WITH
DEPARTMENT OF MECHANICAL ENGINEERING
UNIVERSITY OF TORONTO

THE OXIDATION, IGNITION, AND DETONATION OF FUEL VAPORS AND GASES

XII. THE HIGH COMPRESSION RATIO OTTO CYCLE GAS ENGINE AND THE ADVERSE EFFECT OF HIGH JACKET TEMPERATURES ON THERMAL EFFICIENCY

By R. O. King, Edwin J. Durand, and J. Alex. Morrison

DEFENCE RESEARCH BOARD
Date Rec'd FEB 10 1950
From
Acc. No.
No. of Copies 2
Not For Abstracting

Reprinted from
CANADIAN JOURNAL OF RESEARCH
F, 27 : 435-449. 1949

1507 860

106-04310/

Reprinted from Canadian Journal of Research, F, 27 : 435-449. November, 1949

THE OXIDATION, IGNITION, AND DETONATION OF FUEL VAPORS AND GASES

XII. THE HIGH COMPRESSION RATIO OTTO CYCLE GAS ENGINE AND THE ADVERSE EFFECT OF HIGH JACKET TEMPERATURES ON THERMAL EFFICIENCY[1]

By R. O. King,[2] Edwin J. Durand,[3] and J. Alex. Morrison[4]

Abstract

Town gas was used as the fuel for the C.F.R. engine during a series of engine trials run at a compression ratio of 10 : 1 and at jacket temperatures of 140°, 212°, 302°, and 351° F. The mixture strength at each jacket temperature was varied from the weakest to the richest on which the engine would run steadily. The object of the trials was (1) to determine the effect of jacket temperature on thermal efficiency and (2) to compare the performance of an Otto cycle engine run at 10 : 1 compression ratio with that of a compression ignition (Diesel) oil engine. The results of the trials show that thermal efficiency decreases as the jacket temperature is raised, that is, the decrease more than offsets the gain due to decrease of heat loss during combustion and expansion. It is indicated by the character of the results and by experiments described earlier that the observed loss of efficiency is due to flameless surface combustion of the fuel during compression. The performance of the C.F.R. engine running on town gas at 10 : 1 compression ratio as compared with that of a compression ignition oil engine running at 12 : 1 was superior in respect of maximum power (I.M.E.P.) developed and range of quality control and not greatly inferior in respect of thermal efficiency.

Introduction

It was shown by experiments described in Part VI (3) that town gas, containing hydrogen in large concentration, could be used as fuel for an Otto cycle engine at compression ratios as high as 10 : 1, if precautions were taken to prevent the accumulation in the combustion space of finely divided carbon arising from pyrolysis of the lubricating oil. The power measurements made during the experiments were of a preliminary character but sufficed to show that the increase of compression ratio beyond the 5 : 1 commonly used, made possible by avoidance of nuclear ignition, was accompanied by a commensurate increase in power and thermal efficiency.

The C.F.R. knock testing engine, belt-connected to a d-c. generator, was used for the experiments mentioned. Measurements of power output and friction loss were made consequently by indirect methods and accuracy suffered accordingly. The experimental results of this Part were obtained after a swinging field electric dynamometer had been direct connected to

[1] *Manuscript received May 17, 1949.*
Contribution from Defence Research Board (Canada) in association with the Department of Mechanical Engineering, University of Toronto.
[2] *Research Scientist, Defence Research Board and Honorary Lecturer, Department of Mechanical Engineering, University of Toronto. Lately Director of Scientific Research and Development, Royal Canadian Navy.*
[3] *Lecturer, Department of Mechanical Engineering, University of Toronto.*
[4] *Supervisor of Appliance Laboratory, Consumers' Gas Co., Toronto.*

the engine, and it is considered that an acceptable accuracy of measurement was attained.

The objectives of the experiments were, (1) to determine if flameless surface combustion during compression would more than offset the gain in thermal efficiency due to raising the jacket temperature and (2) to obtain reliable data for an Otto cycle engine running on town gas at a compression ratio of 10 : 1 for a comparison of the performance with that of a compression ignition (Diesel) engine, in respect of power, efficiency, and range of quality control.

Trials were carried out with the jacket coolant at temperatures of 140°, 212°, 302°, and 351° F., mixture strength being varied over the greatest possible range and optimum ignition timing always being used. The temperature of 140° F. can be taken as that of the jacket coolant in motor car engines; 212° F. is the jacket coolant temperature of the A.S.T.M.–C.F.R. method of knock testing; the higher temperatures of 302° and 351° F. represent jacket coolant temperatures of liquid cooled aircraft engines. The cylinders of air cooled engines attain even higher temperatures.

Experimental Arrangements and Results

The C.F.R. engine, which had been used for some weeks for fuel testing, was dismantled and all carbon and other deposits removed from combustion chamber surfaces. The combustion space was maintained sufficiently free of the accumulations of finely divided carbon, which promote pre-ignition and detonation, by the cleaning routine described in Parts V (2) and VI (3).

The valves were refaced and hand lapped into the seats. The piston rings were in good condition and free in the grooves, all of which were clean. The ignition system was checked and the breaker points set to the standard 0.020 in. gap. The spark plug was cleaned and the gap reduced from 0.025 in. to 0.012 in. as had been found necessary in earlier experiments at the high compression ratio of 10 : 1. The setting of the micrometer used for measuring compression ratio was checked, and maximum compression pressures corresponding to compression ratio settings were found to be in agreement with standard C.F.R. values. The maximum compression pressure at 10 : 1 compression ratio was 290 lb. per sq. in., measured during trials at 900 r.p.m. and with the jacket coolant at 212° F.

Lubrication

A commercial grade of lubricating oil, S.A.E. 30, free of additives, was used for engine trials at jacket temperatures of 140° and 212° F. A commercial grade of heavy duty oil, Esso S.A.E. 50, recommended for use with Diesel engines was used for trials made with jacket temperatures of 302° and 351° F.

Jacket Temperatures

The low temperature was obtained by passing tap water through the jacket. The water entered at the lower end of the jacket at 60° F. and flow was

regulated to maintain a nearly constant outlet temperature of 140° F. The standard evaporative method of cooling, using water, was employed to obtain a jacket temperature of 212° F. The higher jacket temperatures of 302° and 351° were obtained by evaporative cooling, using ethylene glycol suitably diluted with water, an extra section being fitted to the reflux condenser.

Power Measurement

The belt driven main and auxiliary generators of the C.F.R. unit were disconnected from the engine. The main generator was then driven as a motor from the laboratory d-c. supply and it in turn drove the auxiliary generator which supplies current for engine auxiliaries, such as the bouncing pin and the ignition system. The engine was then direct connected to a d-c. swinging field dynamometer with beam type scale. Thus, brake horsepower was measured. Indicated horsepower was obtained by adding to the observed B.H.P. that required to overcome friction and pumping losses when fuel was not added to the air supply and the engine "motored" by the dynamometer. "Motoring" loss was measured after every set of power observations as soon as possible after stopping the fuel supply. A set of observations comprised 6 power measurements made to determine optimum ignition timing for any particular rate of gas supply (mixture strength). Average values of the horsepower absorbed by friction and pumping losses, described hereafter as L.H.P., were as below,—

140° F. jacket, S.A.E. 30 oil—1.45 L.H.P.
212° F. jacket, S.A.E. 30 oil—1.31 "
300° F. jacket, S.A.E. 50 oil—1.24 "
350° F. jacket, S.A.E. 50 oil—1.17 "

The relatively great loss at 140° F. jacket temperature is due to the entrance of tap water into the lower end of the cylinder jacket at 60° F.

Measurement of Rate of Gas Consumption

Measurements were made by members of the technical staff of the Consumers' Gas Company of Toronto, using a standard dry meter especially fitted and calibrated for accurate measurement.

Calorific Value of the Town Gas

The composition and calorific value of Toronto town gas are closely controlled. The composition remained constant during the engine trials at jacket temperatures of 212° F. and higher, made on successive days, but differed slightly from that during the trial at 140° F. jacket temperature made some weeks earlier, as shown by the data of Table I.

The higher calorific value of the gas used for the trial at 140° F. jacket temperature, determined by calorimeter in the Gas Company laboratory, was 465.1 B.t.u. per cu. ft. and the corresponding lower calorific value obtained by calculation was 422.2 B.t.u. per cu. ft. Values obtained similarly for the gas when used later at the higher jacket temperatures were 470 and 427

TABLE I

COMPOSITION OF TORONTO TOWN GAS AT TIMES OF ENGINE TRIALS; PERCENTAGE VOLUMES

Constituents	Trial at 140° F., jacket temperature	Trials at higher jacket temperatures
Hydrogen	48.3	47.4
Carbon monoxide	23.3	21.7
Methane	11.0	12.4
Nitrogen	7.1	7.6
Carbon dioxide	3.7	4.2
Ethylene	3.0	3.8
Ethane	1.8	1.3
Benzene	1.1	0.9
Oxygen	0.8	0.7

B.t.u. per cu. ft. The "lower" values were used for calculating indicated thermal efficiency. All are for the gas at 60° F. and 30 in. of mercury.

Carburetor Arrangement

The method of admitting gas to the C.F.R. carburetor as used in preliminary experiments is shown by Fig. 1 of Part VI (3). The open end of the gas admission tube was fully exposed to the air oscillations in the induction system of the engine. The oscillations were probably without effect on the accuracy of the gas meter, but as a precaution the open end of the tube was closed and the gas passed through three rings of small holes in the wall of the tube. The air heater of the C.F.R. engine was left in place and provided a mixing chamber for the gas and air.

Ignition Timing for Maximum Power

The required ignition timing at constant engine speed and compression ratio varies with jacket temperature and mixture strength. Air and gas were both supplied to the engine at atmospheric pressure. The ignition advance required for the maximum power obtainable for a particular rate of gas consumption can be determined only by experiment. Thus the brake horsepower developed when a particular mixture was used was measured at a series of ignition timings giving rising and falling power. The optimum timing was then obtained from a plot of the results.

The method is illustrated by the graphs of Fig. 1, plotted to determine the optimum ignition timing for 14 rates of gas consumption ranging from that required for the weakest to the richest mixtures on which the engine would run steadily. It will be seen by reference to the graphs that optimum ignition timing varies over a wide range, and is somewhat sharply defined at any particular mixture strength. Thus at the weakest mixture on which the engine would run continuously, given by a rate of gas consumption of 47.5 cu. ft. per hr., optimum timing was 44° advance and output was 1.92 B.H.P. The maximum power output of 4.06 B.H.P. was obtained at a rate of gas consumption of 92.5 cu. ft. per hr., and optimum ignition timing was then 13.5° advance. The graphs for mixtures richer than that required for maxi-

mum power are given separately in the right hand top corner of Fig. 1 to save confusion. They show, as usually found, that optimum ignition timing varies less with mixture strength on the rich than on the weak side of the maximum power mixture strength.

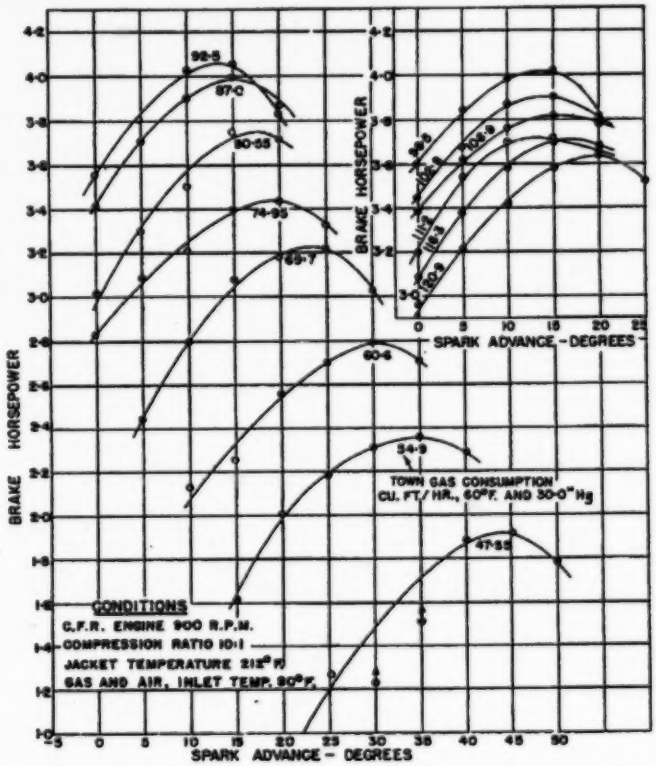

FIG. 1. *Graphs for the determination of optimum ignition timing for each of the 14 rates of gas consumption required for the set of trials made at a jacket coolant temperature of 212° F.*

The experimental results required for the graphs of Fig. 1 were obtained when the jacket coolant temperature was 212° F. Similar sets of graphs were obtained for jacket coolant temperatures of 302° and 351° F. and for a jacket coolant temperature (outlet) of 140° F.

Brake and Indicated Horsepower, Mechanical Efficiency, 212° F. Jacket Coolant Temperature, Optimum Spark Advance

The experimental results given by the graphs of Fig. 1 were used to plot the graphs of Fig. 2. Graph *A* of the figure gives the relation between optimum spark advance and rate of gas consumption and Graph *B* the corresponding brake horsepower. Graph *C*, for indicated horsepower, represents the brake horsepower plus the horsepower (L.H.P.) lost in overcoming mechan-

ical and fluid friction, amounting to 1.31 B.H.P. at the jacket coolant temperature of 212° F. Graph *D*, for mechanical efficiency, gives the percentage relation of brake to indicated horsepower. It will be noted that the value at maximum power output is approximately 76%, a reasonable value in view of the engine's not having been designed for high output and low friction loss.

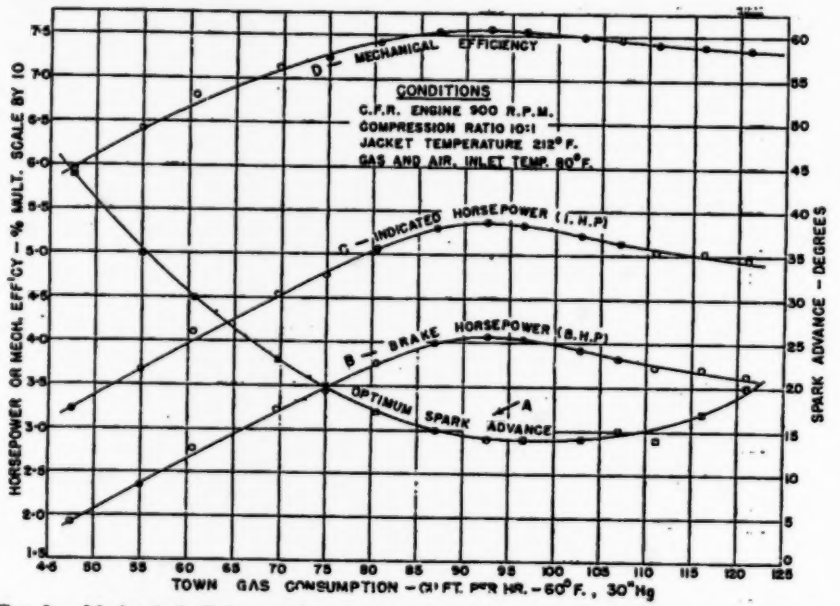
D — MECHANICAL EFFICIENCY
CONDITIONS
C.F.R. ENGINE 900 R.P.M.
COMPRESSION RATIO 10:1
JACKET TEMPERATURE 212° F.
GAS AND AIR INLET TEMP. 80° F.
C — INDICATED HORSEPOWER (I.H.P.)
B — BRAKE HORSEPOWER (B.H.P.)
OPTIMUM SPARK ADVANCE
A
HORSEPOWER OR MECH. EFF'CY — % MULT. SCALE BY 10
SPARK ADVANCE — DEGREES
TOWN GAS CONSUMPTION — CU. FT. PER HR. — 60° F., 30" Hg

FIG. 2. *Mechanical efficiency and brake and indicated horsepower at optimum spark advance—jacket coolant, 212° F.*

Indicated or brake mean effective pressures can be obtained by multiplying values given by Graphs *C* and *B* by the factor 23.4. Thus I.M.E.P. at maximum power output was 125 lb. per sq. in.

Indicated Thermal Efficiency and Power

The internal combustion engine, being noncondensing, is restricted to converting into work some part only of the "lower" heat value of the fuel used. It is necessary therefore to use that value for calculations of thermal efficiency if comparable results for a variety of fuels are to be obtained, because the difference between the lower and higher values varies with the nature of the fuel.

Similarly, when the efficiencies of a variety of engines are compared it is necessary to take into account the total power developed, because the fraction used in overcoming friction varies with design and operating conditions. The total is represented by the indicated horsepower or by the indicated mean effective pressure.

Thus, considering as an example a particular set of engine trials at 900 r.p.m., the jacket coolant at 212° F. and compression ratio 10 : 1, the data used for the calculation of indicated thermal efficiency were as follows,—

Metered rate of gas consumption, 96.6 cu. ft. per hr.
Lower calorific value of the gas at 60° F. (520° R) and 30 in. of mercury barometric pressure, 422 B.t.u. per cu. ft.
Gas temperature at meter, 73.7° F. (533.7° R.)
Barometric pressure, corrected for temperature, 29.33 in. of mercury
Brake horsepower by dynamometer, 4.06
Friction horsepower (L.H.P.) by dynamometer, 1.31
Indicated horsepower by addition, 5.37

The rate of heat input to the engine is then,—

$$422 \times \frac{520}{533} \times \frac{29.3}{30.0} \times \frac{96.6}{60} = 646 \text{ B.t.u. per min.}$$

and the equivalent ft. lb./ per min. $= 646 \times 778$

The rate at which heat is converted into work is
I.H.P. (5.37) $\times$ 33000, ft. lb. / per min.
and per cent indicated thermal efficiency

$$= 100 \times \frac{5.37 \times 33000}{646 \times 778} = 35.2$$

Collected Results of Engine Trials

The results of trials made with a jacket coolant temperature of 212° F. are given by the graphs of Fig. 3. They are plotted on a brake horsepower base extending from the minimum power at which the engine would run steadily to the maximum power that the engine would develop. The obtainable range of B.H.P. is shown by the vertical broken lines. It will be noted that power reaches a maximum for a particular rate of gas consumption and then diminishes as the rate is further increased. Thus the familiar consumption "loop" is obtained.

Indicated thermal efficiency attains relatively high values at weak mixtures, decreases slowly until the mixture strength for maximum power is reached, and then very rapidly with further increase in mixture strength. Then as power decreases at the same time, an indicated thermal efficiency "loop", as shown by the relevant graph of the figure, is obtained.

The results of trials made with a jacket temperature of 351° F. are presented similarly by the graphs of Fig. 4. The similar graphs for trials made at 140° and 302° F. are not reproduced because of space limitation.

The features of special interest in respect of the trials made at the four jacket coolant temperatures of the experiments are exhibited by the graphs of Fig. 5. The *A* graphs are consumption loops, and *B* graphs indicated thermal efficiency loops. They show in a striking way the progressive de-

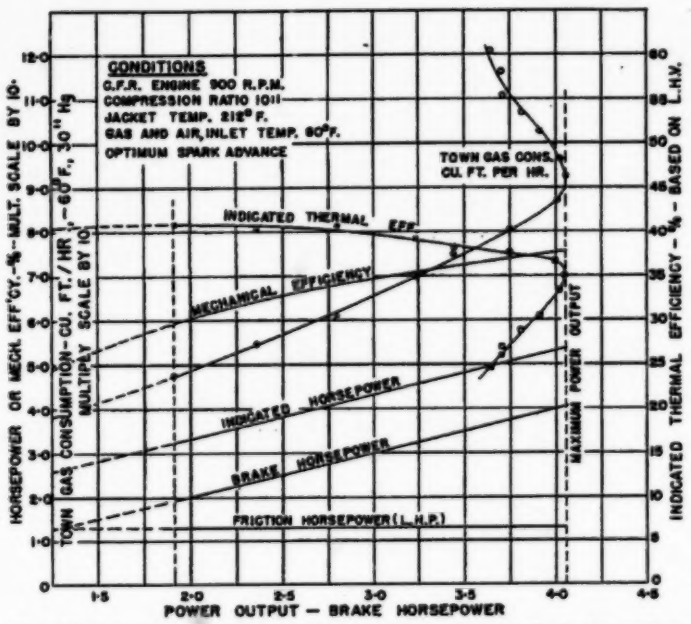
CONDITIONS
C.F.R. ENGINE 900 R.P.M.
COMPRESSION RATIO 10:1
JACKET TEMP. 212° F.
GAS AND AIR, INLET TEMP. 60°F.
OPTIMUM SPARK ADVANCE
TOWN GAS CONS. CU. FT. PER HR.
INDICATED THERMAL EFF.
MECHANICAL EFFICIENCY
INDICATED HORSEPOWER
BRAKE HORSEPOWER
FRICTION HORSEPOWER (I.H.P.)
MAXIMUM POWER OUTPUT
HORSEPOWER OR MECH. EFF'CY. -% - MULT. SCALE BY 10
TOWN GAS CONSUMPTION - CU. FT./HR. - 60°F., 30" Hg. MULTIPLY SCALE BY 10
INDICATED THERMAL EFFICIENCY - % - BASED ON L.H.V.
POWER OUTPUT — BRAKE HORSEPOWER

FIG. 3. *Graphical presentation of the results of trials made with the jacket coolant at 212° F.*

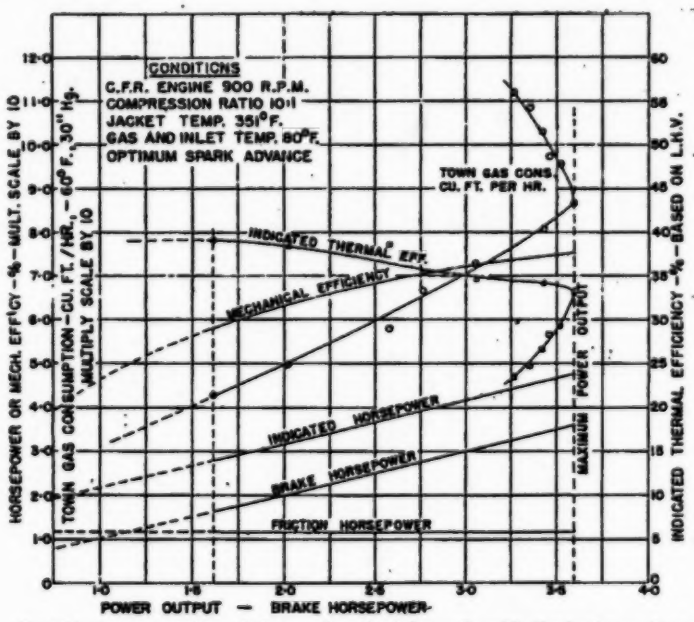
CONDITIONS
C.F.R. ENGINE 900 R.P.M.
COMPRESSION RATIO 10:1
JACKET TEMP. 351°F.
GAS AND INLET TEMP. 60°F.
OPTIMUM SPARK ADVANCE
TOWN GAS CONS. CU. FT. PER HR.
INDICATED THERMAL EFF.
MECHANICAL EFFICIENCY
INDICATED HORSEPOWER
BRAKE HORSEPOWER
FRICTION HORSEPOWER
MAXIMUM POWER OUTPUT
HORSEPOWER OR MECH. EFF'CY. -% - MULT. SCALE BY 10
TOWN GAS CONSUMPTION - CU. FT./HR. - 60°F., 30" Hg. MULTIPLY SCALE BY 10
INDICATED THERMAL EFFICIENCY - % - BASED ON L.H.V.
POWER OUTPUT — BRAKE HORSEPOWER

FIG. 4. *Graphical presentation of the results of trials made with the jacket coolant at 351° F.*

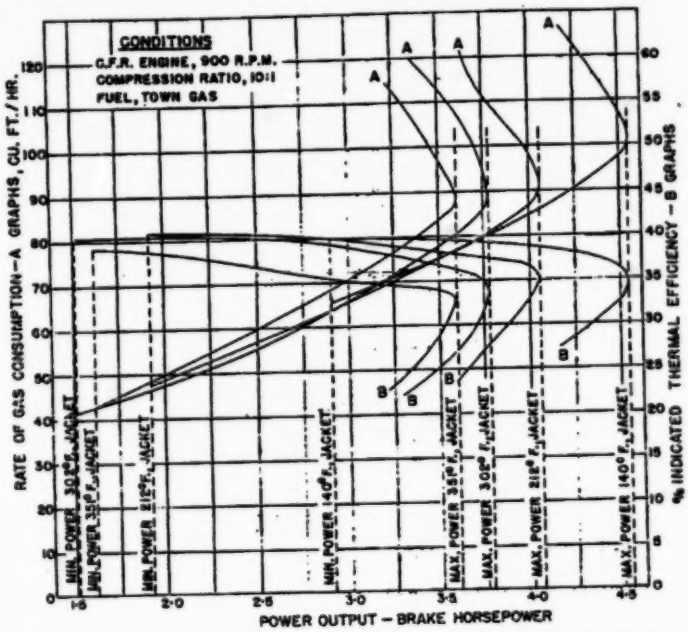
CONDITIONS
C.F.R. ENGINE, 900 R.P.M.
COMPRESSION RATIO, 10:1
FUEL, TOWN GAS
RATE OF GAS CONSUMPTION—A GRAPHS, CU. FT./HR.
% INDICATED THERMAL EFFICIENCY—B GRAPHS
POWER OUTPUT—BRAKE HORSEPOWER
A
B
MIN. POWER 302° F. JACKET
MIN. POWER 351° F. JACKET
MIN. POWER 212° F. JACKET
MIN. POWER 140° F. JACKET
MAX. POWER 351° F. JACKET
MAX. POWER 302° F. JACKET
MAX. POWER 212° F. JACKET
MAX. POWER 140° F. JACKET

FIG. 5. *Consumption loops given by A graphs and indicated thermal efficiency loops by B graphs; for trials made with jacket coolant temperatures of 140°, 212°, 302°, and 351° F.*

crease of indicated thermal efficiency and power as jacket coolant temperature is raised from 140° to 351° F. Maximum and minimum powers obtained for each jacket coolant temperature are shown by the vertical broken lines. It will be noted that a relatively high value of minimum power was obtained when the jacket coolant temperature was 140° F. This anomalous result is thought to be due to ignition failure at weaker mixtures.

Summary and Discussion of Experimental Results

It was not considered possible, prior to the experiments described in Parts V (2) and VI (3), to operate an Otto cycle engine on hydrogen or town gas at a compression ratio of 10 : 1, except when using weak mixtures and consequently obtaining relatively low values of mean effective pressure. Thus, there are not extant any engine trial results for Otto cycle engines with which the experimental results given in this Part can be compared. Recourse must therefore be to the trial results of the compression ignition type of the Diesel engine. The most comprehensive described trials of that type of engine are, so far as known, those by H. B. Taylor (8, pp. 35-85). Data are given for trials run at 12 : 1 compression ratio and speeds of 800 and 1000 r.p.m. and with injection timing fixed or advanced progressively with decrease in mixture

strength. A comparison of the C.F.R. Otto cycle engine trial results with those given by Taylor for the compression ignition engine are therefore of interest although weighted in favor of the latter by the higher compression ratio of 12 : 1 at which they were run. The temperature of the jacket coolant is not mentioned in Taylor's paper but can be assumed to have been that generally used, namely, 60° C. (140° F.).

Thermal Efficiency as Affected by Jacket Temperature

An increase in jacket temperature, by diminishing heat loss during the combustion and expansion periods, should give rise to a small increase in thermal efficiency. The possibility that the increase might be more than offset by the adverse effect of surface oxidation of the fuel has not heretofore been considered, the established (textbook) view being that "no significant surface reaction occurs in the engine" (1, p. 2934).

The adverse effect of flameless surface combustion on the power developed by an Otto cycle engine running on gasoline was shown by the experiments described in Part XI (4). The experiments of this Part were made to determine if a similar effect would be obtained in suitable conditions if the engine were run on town gas containing a large proportion of hydrogen, which reacts with metallic oxide at relatively low temperatures to form steam. The experimental results obtained accordingly are summarized in Table II. The data given for jacket coolant temperatures of 212° F. are from the graphs of Figs. 2 and 3. The data relating to other jacket coolant temperatures are from similar graphs, which are not reproduced because of space limitations.

TABLE II

INDICATED THERMAL EFFICIENCY AND INDICATED MEAN EFFECTIVE PRESSURE AS AFFECTED BY JACKET COOLANT TEMPERATURE

Jacket coolant temperature, °F.	Max. power		75% max. power		Min. power	
	I.M.E.P., lb./sq. in.	I.T.E., %	I.M.E.P., lb./sq. in.	I.T.E., %	I.M.E.P., lb./sq. in.	I.T.E., %
140	140.4	35	105	40	(102)	(40)
212	125.0	35	94	40	76	41
302	117.0	33.5	88	40	65	40
351	111.0	32.5	83	37	65	39

With reference to Table II, consider first the effect on thermal efficiency of raising the jacket temperature from 140° to 212° F. The estimates by Ricardo (6, p. 87) and Ricardo and Glyde (7, p. 103), which are not based on experiment, led to the conclusion that an increase of from 0.4% to 1.0% would be obtained. This no doubt is an accurate estimate, taking into account heat loss only. The *experimental* results given in Table II show, however, that no change in thermal efficiency occurred within the jacket coolant temperature range 140° to 212° F., the minimum power measurements at 140° F. being disregarded, for the reason already mentioned. It may be

concluded accordingly that the gain due to the lower heat loss during combustion and expansion was just equal to the loss due to surface oxidation of the fuel during compression.

Consider now the effect on efficiency of raising the jacket coolant temperature from 212° to 351° F. There should, of course, be a gain in thermal efficiency due to lower heat loss to the jacket but instead efficiency decreases by 7% at maximum power and by 7½% at three-quarter power. The total loss attributable to surface oxidation would therefore be the sum of the percentages mentioned plus whatever gain there might have been, taking into account the lower heat loss to the jacket.

The smaller efficiency loss of 4.9%, observed when using the weak mixture required for minimum power, is attributed to the decrease in velocity of the surface reaction with decrease in mixture strength.

Thermal Efficiency, Weak Mixtures

Thermal efficiency should, according to generally accepted theory, approach the Air Standard value as the concentration of fuel in the mixture with air approaches zero. The experimental results obtained with the C.F.R. engine running at 10 : 1 compression ratio indicate, on the contrary, that thermal efficiency attains a maximum value at a mixture strength richer than the weakest on which the engine continues to run. A similar effect was obtained as a result of earlier experiments with town gas [Part VI (3)]. The observed decreases from the maximum, although small, are of sufficient interest to justify further experiments.

Quality Control

A feature of the Diesel or compression ignition type of Diesel is that power output at any particular speed can be varied over a wide range by varying the rate of fuel supply, that is, the "quality" of the fuel–air mixture. It became of interest, therefore, to determine the degree of quality control obtained when the C.F.R. engine was run at 10 : 1 compression ratio and with jacket coolant temperature varied over a wide range.

Percentage quality control is taken to be:—

$$\frac{\text{Maximum B.H.P.} - \text{Minimum B.H.P.}}{\text{Maximum B.H.P.}} \times 100$$

Relevant data from the C.F.R. engine experiments are set out in Table III.

TABLE III

QUALITY CONTROL DATA, C.F.R. ENGINE
10 : 1 COMPRESSION RATIO

Temperature of jacket coolant, °F.	Max. B.H.P.	Min. B.H.P.	Friction H.P.	Percent *Q* control
140	4.55	2.91	1.35	36
212	4.05	1.91	1.31	53
302	3.76	1.52	1.24	60
351	3.60	1.61	1.17	55

The relatively low percentage quality control at 140° F. jacket coolant temperature is believed to be due to ignition failure, as already mentioned.

The obtaining of a wide range of quality control depends on continuing ignition as mixture strength is progressively weakened. The spark ignition system of the C.F.R. engine is not especially adapted for use with very weak mixtures, and although the results are somewhat irregular, as would be expected, it is of interest to compare them with those relating to compression ignition engines as given by Taylor (8).

The percentage quality control obtainable with the compression ignition engine was found by Taylor (8) to diminish with speed and to be little affected by injection timing. Relevant data, taken from page 47 of the reference quoted, are given in Table IV.

TABLE IV

QUALITY CONTROL DATA, COMPRESSION IGNITION ENGINE, 12 : 1 COMPRESSION RATIO

Engine r.p.m.	Max. B.H.P.	Injection degrees advance	Min. B.H.P.	Injection advance	Percent Q control
800	64.6	11.5	30		54
800	65.0	12.5	30	31.5	54
1000	80.0	12.0	40		50
1000	79.5	12.0	40	29.5	50

The percentage quality control for the compression ignition engine if run at 900 r.p.m. can be taken as 52. The Otto cycle C.F.R. engine run at that speed and at the lower compression ratio of 10 : 1 affords quality control of 53, 60, and 55% for jacket temperatures of 212°, 302°, and 351° F. respectively.

Quality control is determined by the spread between maximum and minimum brake horsepower (B.M.E.P.). The minimum cannot be greater than the B.H.P. required to overcome friction and pumping losses and may be more nearly reached with the Otto cycle engine by improving spark ignition at weak mixtures. The maximum can be the greater for the Otto cycle engine because of the inability of the compression ignition engine effectively to utilize the mixture strength required for maximum power.

Volumetric Efficiency, Spark Advance

The engine was always run at constant speed and compression ratio, and any decrease in volumetric efficiency observed on raising the jacket temperature would be due to the consequent heating of the induced charge. The extent of the decrease for a particular increase of jacket temperature would depend mainly on the turbulence and thermal conductivity of the gas–air mixture entering the cylinder. The mixture enters the cylinder of the C.F.R. with a high degree of turbulence because of high velocity attained in passing through a somewhat restricted inlet valve opening, and the gas–air mixture

containing hydrogen in large proportion is of relatively high thermal conductivity. The conditions of the engine trials were therefore such that a large decrease in volumetric efficiency would be expected. Moreover, the increase in charge temperature due to increasing the jacket temperature would give rise to an increase in compression temperature, a corresponding increase in rate of flame propagation after spark ignition, and a consequent decrease in optimum spark advance. Relevant data collected from the experimental results are set out in Table V.

TABLE V

EFFECT OF JACKET TEMPERATURE ON VOLUMETRIC EFFICIENCY AND OPTIMUM SPARK ADVANCE

Jacket coolant temperature, ° F.	Maximum I.H.P.	Gas consumption cu. ft./hr.	Spark advance
140	6.00	102	16°
212	5.40	91.5	14°
302	5.00	88.5	8°
351	4.75	87	4°

When the engine charge is heated during induction, volumetric efficiency diminishes and indicated power decreases in the same proportion if speed and compression ratio remain constant, according to a discussion by Pye (5, p. 240), of experiments by Ricardo. On this basis the decrease of maximum I.H.P. from 6.00 to 4.75, Table V, indicates a large decrease of 20.8% in volumetric efficiency. The method of calculation is based necessarily on the generally accepted view that indicated thermal efficiency does not *diminish* with increase of jacket temperature. It has been shown, however, Table II, that a decrease of 7% was obtained on increasing jacket coolant temperature from 140° to 351° F. Thus of the total decrease of 20.8% in I.H.P., 13.8% represents that due to decrease of volumetric efficiency.

The change in volumetric efficiency can be calculated by another method. Thus, when the engine charge is gas and air, both being supplied at the same temperature and pressure, the rate of consumption of either is proportional to volumetric efficiency *if mixture strength remain constant.* It may safely be assumed that the mixture strength for maximum power is invariable. With reference to Table V, rate of gas consumption at maximum power diminishes from 102 to 87 cu. ft. per hr., that is, by 14.7% when the jacket coolant temperature is raised from 140° to 351° F. The decrease in volumetric efficiency of 14.7% agrees as well as could be expected with the value of 13.8% deduced from observed changes in power and efficiency.

Optimum Ignition Timing

The decrease in optimum spark advance as jacket temperature is raised is given in Table V and reflects the increase in rate of flame propagation as compression temperature increases with increase in the temperature of the gas–air mixture at the beginning of compression. The result, so far as engine

operation is concerned, is that the usual effects of excessive spark advance, namely, loss of power accompanied by rough running and knock, are obtained for a relatively small advance beyond the optimum value.

Concluding Comment

The Adverse Effect of Surface Combustion on Thermal Efficiency

The decrease of thermal efficiency obtained on raising the temperature of the jacket coolant of the C.F.R. engine is attributed to the effect of flameless combustion of hydrogen on oxide coated surfaces, to produce steam during compression. The decrease of efficiency becomes greater as mixture strength increases because the velocity of the surface reaction increases accordingly. Thus the decrease in indicated thermal efficiency, Table II, is 7% at the mixture strength for maximum power. It is 10.3%, that is, 50% greater for the richer mixture obtained on increasing rate of gas consumption by 20%. The adverse effect on thermal efficiency due to surface oxidation is similar to that described in Part XI (4), obtained when surface oxidation of liquid fuel vapor to steam and carbon dioxide occurred on a metal surface maintained in an oxidizing atmosphere by continuous decomposition of a metallic antiknock.

It is to be noted that the conclusion in respect of the cause of the decrease in thermal efficiency with increase in jacket temperature is based on the somewhat indirect evidence afforded by earlier experiments, mainly those described in Part XI (4). Further experiments made in a manner to eliminate surface combustion would therefore be of interest.

Otto Cycle Compared with Compression Ignition (Diesel) Performance

Combustion in the compression ignition (C.I.) engine leaves unused from 20 to 25% of the air in the cylinder charge. This is approximately the percentage of the air displaced when town gas in "correct" proportion with air is used as the fuel for an Otto cycle engine. Combustion of the fuel, especially when rich mixtures are used, is the more nearly complete in the Otto cycle engine because of the homogenity of the mixture with air. It would be of interest to compare performances on the basis of the same compression being used. A comparison can however be made at present on the basis of 10 : 1 compression ratio only and 900 r.p.m. for the C.F.R. Otto cycle engine and 12 : 1 compression ratio and 1000 r.p.m. for the C.I. engine. The data for the "best" performance of the C.I. engine used by Taylor (8) are given (page 47 of the reference). The maximum cylinder pressure was 800 lb. per sq. in. and injection timing was advanced as load was reduced.

The maximum I.M.E.P. obtained by Taylor was 133.7 lb. per sq. in., with an indicated thermal efficiency of 38.3%. The corresponding figures for the C.F.R. Otto cycle engine were 140.4 and 35%.

The I.M.E.P. obtained by Taylor at the weakest usable mixture strength was 77.5 lb. per sq. in. with an indicated thermal efficiency of 46.4%. The

corresponding figures for the C.F.R. engine were (at 212° F. jacket temperature) 76 lb. and 41%.

The C.F.R. Otto cycle engine was somewhat superior in respect of the important feature of quality control, as shown by the data of Tables III and IV.

Acknowledgments

The experimental work was carried out with the co-operation of Prof. E. A. Allcut, Head of the Department of Mechanical Engineering, University of Toronto, and Assistant Professor W. A. Wallace. Special acknowledgment is made of the assistance of Mr. George Southall of the laboratory staff of the Consumers' Gas Company, who carried out the long series of gas supply measurements. Assistance was also given in the course of the experiments by Messrs. C. C. Li, E. H. Dudgeon, W. E. Morley, and H. W. M. Mar, all members of the staff of the Department of Mechanical Engineering.

References

1. BEATTY, H. A. and EDGAR, G. *In* The science of petroleum Vol. 4, *Edited by* A. E. Dunstan, Oxford University Press, London, New York, Toronto. 1938.
2. KING, R. O., WALLACE, W. A., and MAHAPATRA, B. Can. J. Research, F, 26 : 264-274. 1948.
3. KING, R. O., WALLACE, W. A., and MAHAPATRA, B. Can. J. Research, F, 26 : 366-373. 1948.
4. KING, R. O. and DURAND, E. J. Can. J. Research, F, 27 : 347-358. 1949.
5. PYE, D. R. The internal combustion engine. 2nd ed. Oxford University Press, London. 1937.
6. RICARDO, H. R. The high speed internal combustion engine. Blackie & Son Ltd., London and Glasgow. 1931.
7. RICARDO, H. R. and GLYDE, H. S. The high speed internal combustion engine. Interscience Publishers, Inc. New York. 1941.
8. TAYLOR, H. B. Proc. Inst. Automobile Engrs. (London), 22 : 35-85. 1927-28.